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COMPARISON OF EXPERIMENTAL AND IDEAL LEAKAGE FLOWS THROUGH LABYRINTH SEALS FOR VERY SMALL PRESSURE DIFFERENCES

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16. Abstract

Leakage flows for stationary straight-through and stepped labyrinth seals were obtained experimentally with room-temperature air for pressure differences of 0.2 to 1.4 inches of water (50 to 350 N/m^2) and pressure ratios of 0.9990 to 0.9999. The analytical leakage prediction method of Egli was shown to be valid for the range of test conditions. Leakage flow increased about 5 percent through both test seals when eccentricity ratio was changed from 0 to 0.72. The straight-through seal was insensitive to flow direction. Leakage flow for the stepped seal was 7 percent greater with flow over the step than with flow against the step.

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SUMMARY

A requirement for handling accurately metered quantities of cooling air in a NASA seal design application generated a need for the determination of seal leakage flows at small pressure differences (0.2 to 1.4 in. of water, 50 to 350 N/m^2), and at pressure ratios of about 0.9990 to 0.9999. An experimental investigation was made to determine the leakage characteristics of three-stage straight-through and stepped labyrinth seals. The investigation was made at nonrotating conditions, using room-temperature air. Leakage flow in both directions through the seals and the effect of eccentricity were studied.

The analytical method developed by Egli proved to be valid for very small pressure differences. Based on the overall pressure difference, the leakage flows for both the straight-through and the stepped seals increased by about 5 percent when the seal position was changed from an eccentricity ratio of 0 to 0.72. Leakage flow for the straight-through seal was insensitive to the direction of airflow. Leakage flow for the stepped seal was 7 percent greater when the flow direction was over the step than when the flow direction was against the step.

INTRODUCTION

Labyrinth seals are widely used in rotating machinery to restrict or control the flow of fluids between adjacent chambers that are at different pressure levels. Studies of leakage through multiple-element labyrinth seals (refs. 1 to 4) have covered a wide range of pressure differences across the seals and a wide range of seal geometries. However, the applicability of these studies to seals operating at very small pressure differences were not obvious. With a requirement for a labyrinth seal operating at pressures as high as 100 psia (6.9×10^5 N/m^2) and at pressure ratios as high as 0.9999, a study of the leakage characteristics was instituted.

In addition to pressure difference, the major factors affecting leakage flow rate of labyrinth seals are the number, thickness, axial spacing, and radial clearance of the knife-edges. The various flow equations evolved to predict leakage flow rates (refs. 1 to 4) were based on specific geometric configurations and certain pressure ratio limitations. The geometry of the present seal was most closely approximated by the study of reference 1. Therefore, a modified version of the analytical method was chosen as a means of correlating leakage flow rates.

The purpose of this investigation was twofold. The first was to determine experimentally if the modified version of the analytical method is valid for predicting seal leakage flows with very low pressure differences. The second was to investigate the effect of eccentricity and flow direction on leakage through a labyrinth seal. To fulfill these objectives, two test seals were studied in a stationary (nonrotating) apparatus at room-temperature conditions. Both test seals contained three stages. One was a straight-through seal (constant-diameter land); the other had a single radial step in the land. Concentric and eccentric positions of the land with respect to the knife-edges and leakage flows in both directions through the seals were studied. The experimentally determined leakage rates were obtained at room temperature (75° F, 24° C), with air pressures ranging from 30 to 100 psia (2.1×10^5 to 6.9×10^5 N/m^2) and for pressure differences across the center knife-edge ranging from 0.2 to 1.4 inches of water (50 to 350 N/m^2).

SYMBOLS

A	area of throttling
b	step to knife-edge distance
e	radial eccentric displacement
g	gravitational acceleration constant
h	height of step in seal land
n	number of knife-edges in labyrinth seal
P	absolute pressure
ΔP	pressure difference
R	gas constant
s	axial spacing between adjacent knife-edges

T absolute temperature of air

$\dot{W}$ weight flow rate of air

α flow coefficient

γ carryover correction factor

Δ thickness of seal knife-edge

δ radial clearance between knife-edge and land for concentric seal position

ϵ eccentricity ratio, e/δ

Subscripts:

d downstream

exp experimental

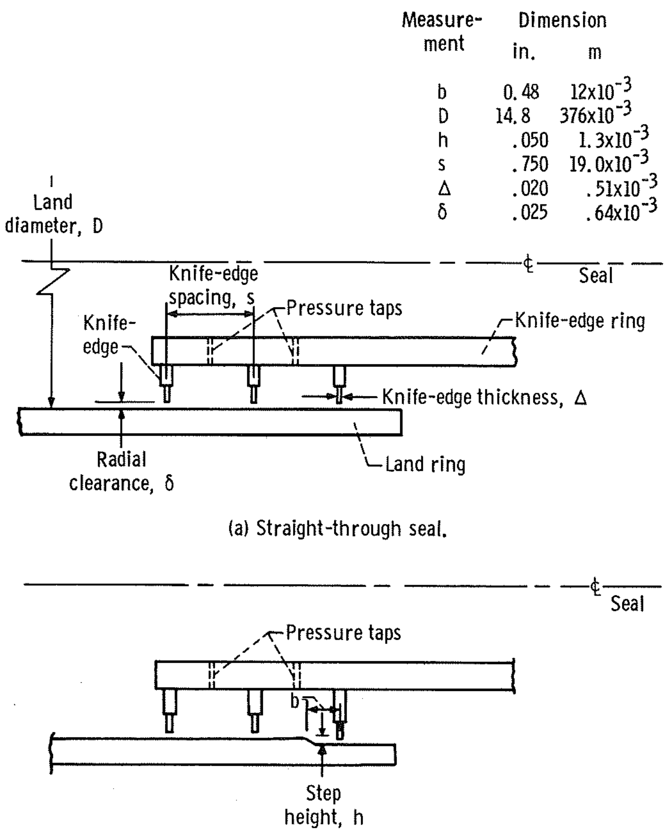
id ideal

u upstream

TEST SEALS

The test seals are shown schematically in figure 1. Figure 1(a) shows the straight-through (constant-diameter land) seal; figure 1(b) shows the seal with a single radial step in the land. This step was located in an axial position between two of the seal knife-edges. This second seal will be referred to herein as a stepped seal. Each test seal had three stages. The knife-edges were on the inner part, and the land was on the outer part of the seals. In this study, both the land and the knife-edges were stationary (nonrotating). In most practical labyrinth seal applications, one of these two components would rotate while the other remained stationary.

At the inception of this study, the exact geometry of the seal for the design application which generated the need for this study had not been established. However, a decision to use a balanced-pressure labyrinth seal for this application had been made. (A balanced-pressure seal is defined as one in which a pressure difference of zero is maintained across the knife-edges of the seal.) Also the monitoring of pressure difference would be done across an internal knife-edge of the labyrinth seal, as shown by the indicated pressure tap locations of figure 1(a). This latter decision was based on the fact that, during the operation of the machinery on which the design application seal would be mounted, the chambers outboard of the seal on either side could be subjected to unequal and unpredictable airflow influences. Such influences could cause relatively



(b) Stepped seal.

Figure 1. - Schematic of test labyrinth seals.

large changes in the indicated pressure differences across the seal compared to the magnitude of the actual pressure difference across the seal.

Previous studies on labyrinth seals, such as reference 1, have shown that a stepped seal was generally superior in performance to a straight-through seal. But the presence of steps in the land (fig. 1(b)) will introduce flow disturbances that could mask the very small pressure drops between adjacent seal chambers that would exist very near balanced-pressure conditions. Furthermore, the magnitude of these disturbances would probably differ with the direction of leakage flow through the seal. Another concern associated with measuring very small pressure differences on the stepped seal, but not investigated in this study, was the change in pumping effect due to a change of radius on the seal land surface. Although this change in radius is small, at high rotational speeds the effect on pumping action can be significant in comparison to the measured pressure differences.

In contrast to a stepped seal, a straight-through labyrinth seal design (fig. 1(a)) could provide identical adjacent chambers that have the same flow characteristics regardless of the direction of leakage flow through the seal. The constant diameter of the

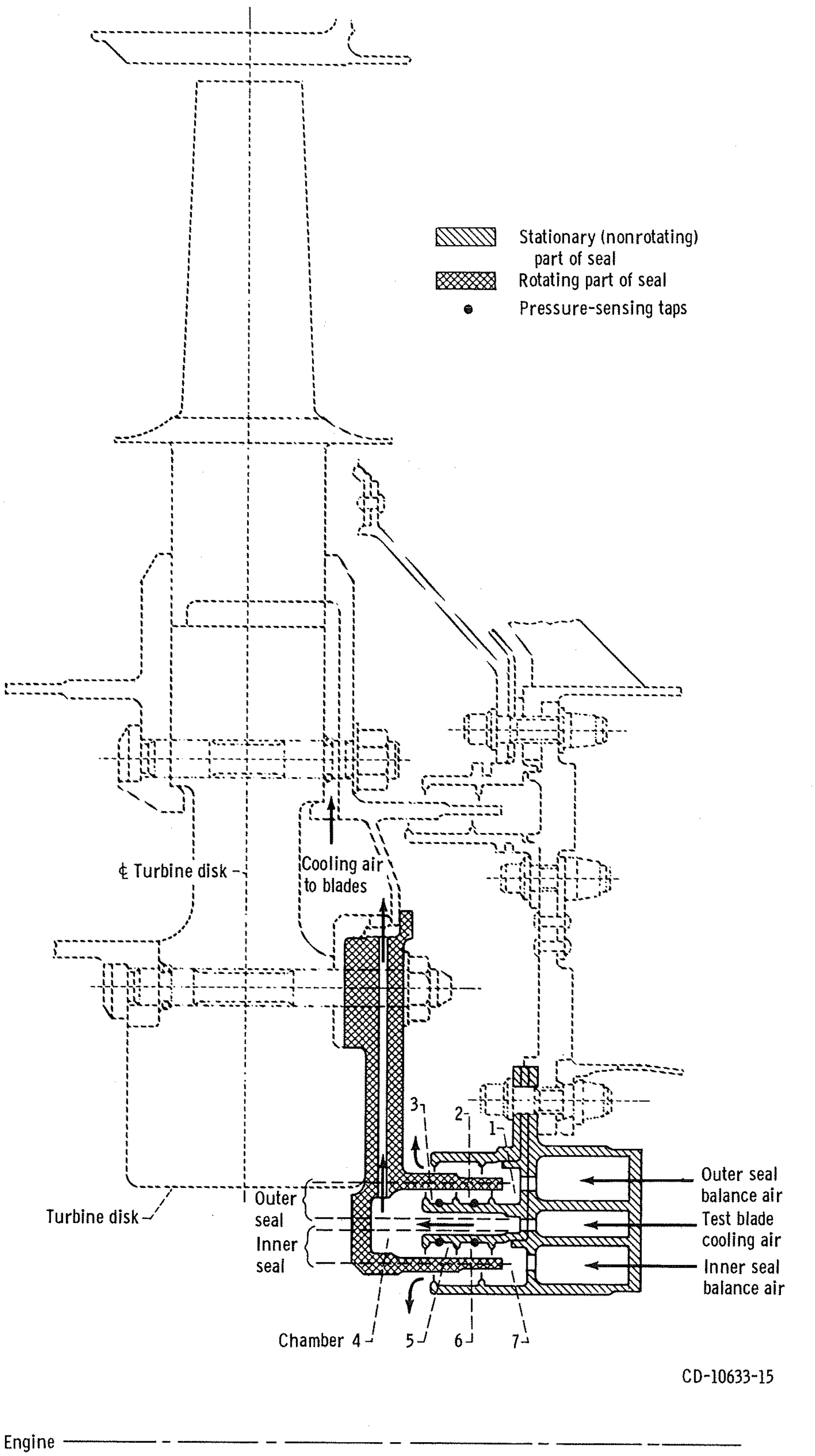


Figure 2. - Cross-sectional view of balanced-pressure labyrinth seal system for turbine cooling research engine.

land would also avoid differences in pumping effect over the axial length of the seal. Against these favorable factors for the straight-through seal must be weighed the inherently larger leakage flow of this type of seal.

In the experimental phase of this study the effects of many of the factors discussed in the preceding paragraphs with regard to the straight-through and stepped labyrinth seals were evaluated. The two test seals were identical in all features with the exception of the step in the land of the stepped seal. Prior to the conclusion of this study, the decision was made to use the straight-through seal in the design application that initiated this study. A detailed description of this design application is given in the appendix. The specific seal is shown in figure 2.

TEST APPARATUS

The test apparatus (fig. 3) consisted of a pressure tank and an air supply system. The test seals were installed in the pressure tank for stationary (nonrotating) testing at room-temperature conditions.

Pressure Tank

The pressure tank is shown schematically in figure 4. The tank was a right circular cylinder approximately 24 inches (0.61 m) in diameter and 5 inches (0.13 m) in height. The parallel base and cover and the cylindrical side of the tank were separate pieces. At assembly, the base and cover were held in position by tie-bolts and spacers. The side was sealed to the base and cover by O-rings installed in circumferential grooves in the edges of these flat plates. Several screws installed in a radial direction anchored the side to the base. Inlet ports for airflow were located at the center and near the outer radius of the base. Similarly, outlet ports for airflow were located at the center and near the outer radius of the cover.

The two test seals could be mounted interchangeably in the pressure tank. The knife-edge ring was mounted on the cover of the tank; the land ring was mounted on the base of the tank. The mounting of the knife-edge ring on the tank cover was such that the ring could be shifted in a radial direction to produce eccentricity. The partially assembled pressure tank with the parts of the stepped test seal in place is shown in figure 5.

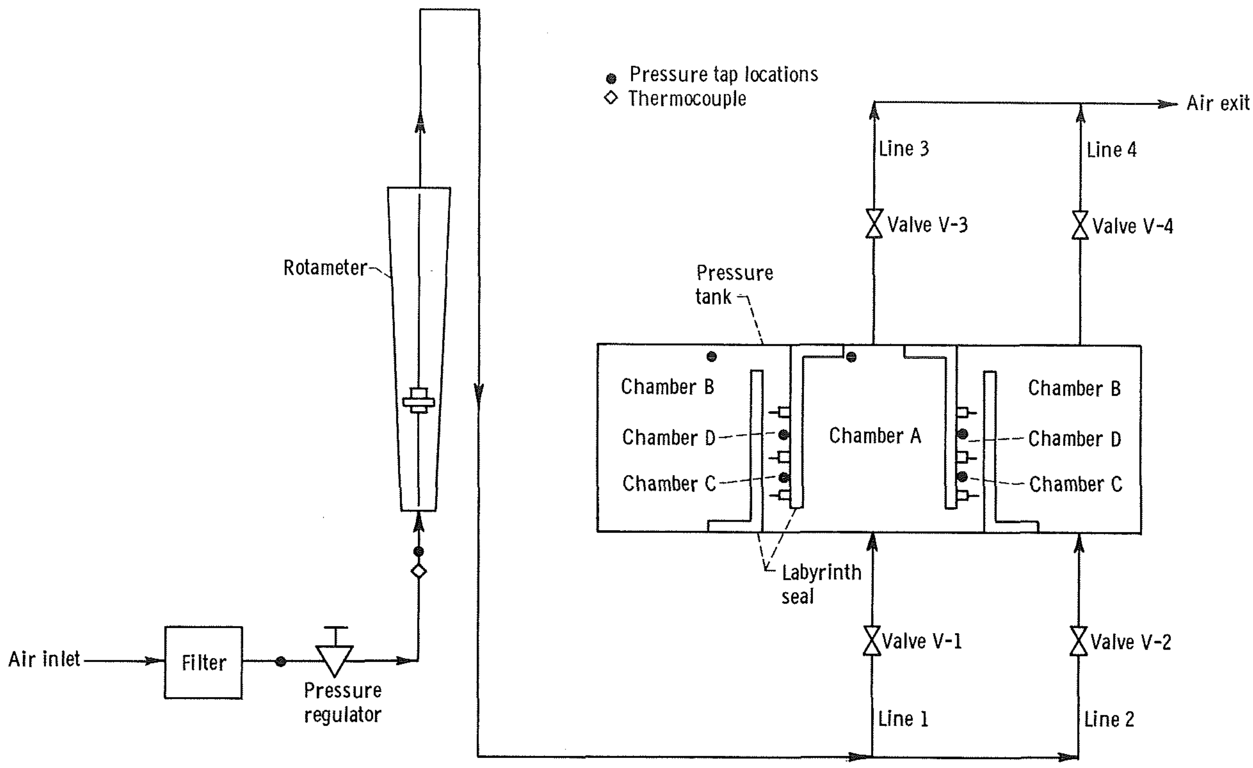
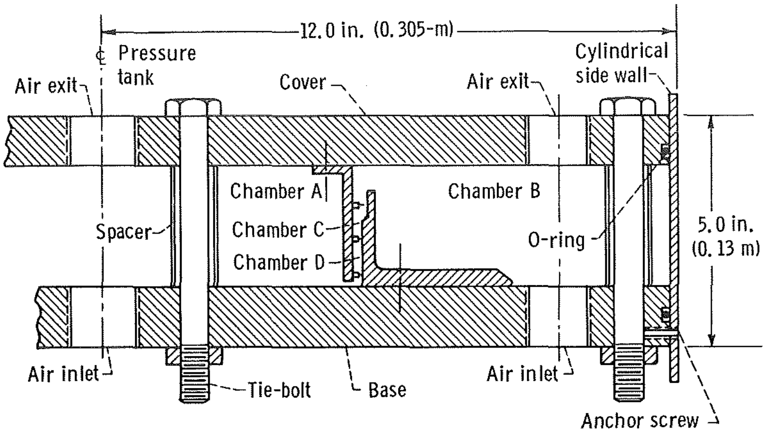


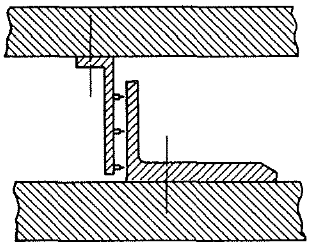
Figure 3. - Schematic of test apparatus for seal leakage flow tests.

Air Supply System

The airflow diagram for the labyrinth seal leakage tests is shown in figure 3. Laboratory air at 75° F (24° C) and at a supply pressure of 125 psig (8.6×10^{5} N/m^2) was introduced through a filter and a pressure regulator to the inlet of the rotameter which was used to meter seal leakage flow. Downstream of the rotameter, the air supply line branched into two inlet lines which lead to the base of the pressure tank. Two air lines attached to the cover of the tank served as exits. A total of four airflow control valves, one in each of the inlet and exit lines, were used. By choosing inlet and exit lines from separate tank chambers, the air entering the tank through one of the inlet lines would always have to pass through the seal to reach the exit line. The airflow direction from chamber A to chamber B is referred to in this report as the normal flow direction. Similarly, the airflow direction from chamber B to chamber A is referred to as the reverse flow direction. Air pressure in the tank and the quantity of airflow through the tank (and thus through the seal) were controlled by the combined manipulation of the pressure regulator and the inlet and exit control valves.



(a) Pressure tank with stepped labyrinth seal installed.



(b) Detail of straight-through labyrinth seal installed in pressure tank.

Figure 4. - Cross-sectional schematic of static test pressure tank with test seals installed.

Instrumentation

The test apparatus was instrumented to measure seal leakage flow rate, air pressure levels, and pressure differences between various locations on the test seals. Seal leakage flow rate was measured by means of a rotameter located in the air supply line upstream of the pressure tank (see fig. 3). Temperature and pressure of the air were measured at the inlet to the rotameter. Pressure level of the air was measured also in chambers A and B in the pressure tank. All measurements of pressure level were made on calibrated precision pressure gages. The pressure differences across the center knife-edge of the test seal were measured by pairs of pressure taps at four locations equally spaced around the circumference of the seal in chambers C and D (see figs 3 and 6). These pressure taps were located midway between the knife-edges and were flush with the surface of the knife-edge ring. The pressure difference across the seal assembly was measured by one pair of pressure taps located across chambers A and B (fig. 3). Each pair of pressure taps was connected to a water manometer.

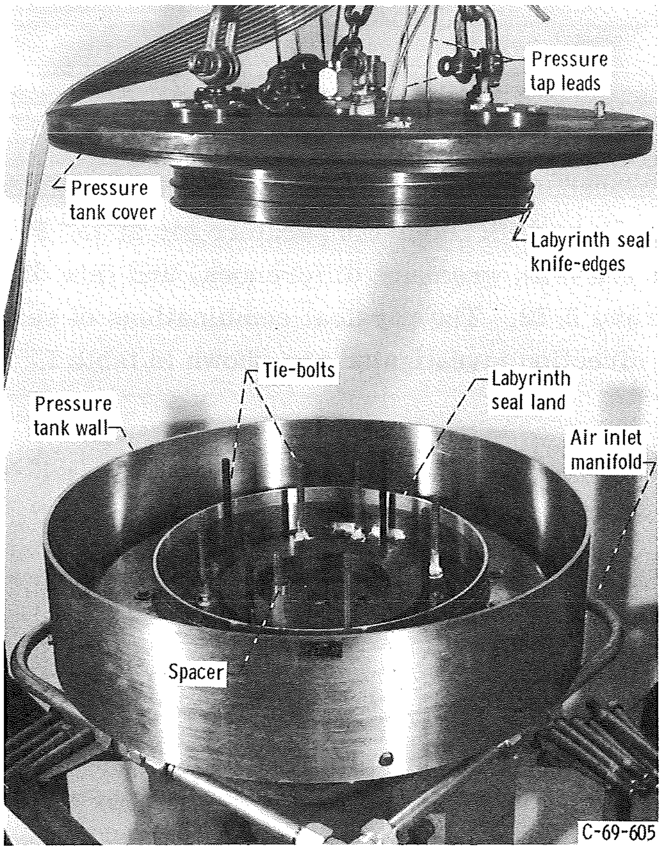


Figure 5. - Static test pressure tank.

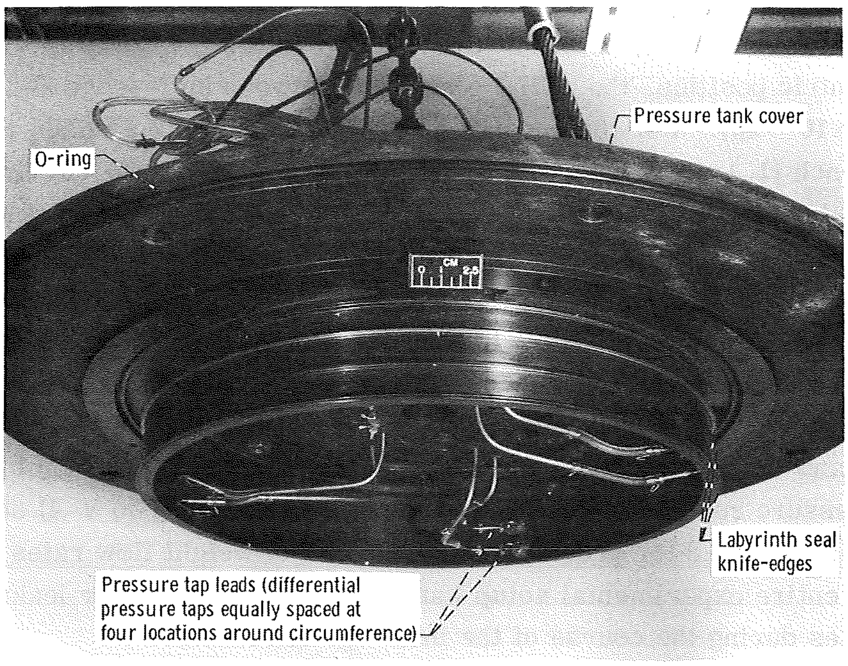


Figure 6. - Labyrinth knife-edge ring mounted on pressure tank cover.

PROCEDURES

Experimental Procedure

The experimental procedure followed in this investigation was designed to determine nonrotating seal leakage flows with room-temperature air at 75° F (24° C) at various combinations of pressure levels, pressure differences, and flow directions, and at eccentricity ratios of 0 and 0.72. The physical combinations of seal configuration, seal position, and flow direction investigated are shown in table I.

TABLE I. - VARIOUS COMBINATIONS OF SEAL CONFIGURATION, POSITION, AND FLOW DIRECTION TESTED

Seal configuration	Seal position	Flow direction
Straight-through	Concentric	Normal
Straight-through	Concentric	Reverse
Straight-through	Eccentric	Normal
Straight-through	Eccentric	Reverse
Stepped	Concentric	Normal
Stepped	Eccentric	Normal
Stepped	Eccentric	Reverse

In the concentric position, the radial clearance between knife-edge and land was 0.025 inch (0.64×10^{-3} m). For the eccentric seal position, the maximum radial clearance was 0.043 inch (1.1×10^{-3} m) and the minimum clearance was 0.007 inch (0.18×10^{-3} m), which resulted in an eccentricity ratio ϵ of 0.72. As explained previously, the normal flow direction occurred when air flowed from chamber A to chamber B (fig. 3); the reverse flow direction occurred when air flowed from chamber B to chamber A. For each combination listed, tank pressures from 30 to 100 psia (2.1×10^{5} to 6.9×10^{5} N/m^2) and pressure differences across the center knife-edge (between chambers C and D) from 0.2 to 1.4 inches of water (50 to 350 N/m^2) were investigated. The pressure levels and the pressure differences were established by the combined manipulation of the pressure regulator and the flow control valves (V-1 to V-4) shown in figure 3. Because of the need for precise measurements of weight flow rates and pressure differences, the entire experimental setup was thoroughly checked for leaks under pressure several times during the course of the investigation.

Calculation Procedure

An analytical method was developed by Egli (ref. 1) to predict leakage flow rates through labyrinth seals. By introducing the equation of state, the Egli equation can be written in a modified form as

$$\frac{\dot{W}\sqrt{T}}{P_u} = (\alpha\gamma)_{id} A \sqrt{\left(\frac{g}{R}\right) \frac{1 - \left(\frac{P_d}{P_u}\right)^2}{n + \ln\left(\frac{P_u}{P_d}\right)}} \tag{1}$$

The term $(P_d/P_u)^2$ in equation (1) can be written as $[(P_u - \Delta P)/P_u]^2$ and expanded into

$$\left(\frac{P_d}{P_u}\right)^2 = \frac{P_u^2 - 2(P_u)(\Delta P) + (\Delta P)^2}{P_u^2} \tag{2}$$

For pressure ratios very close to 1.0, the terms $\ln(P_u/P_d)$ in equation (1) and $(\Delta P/P_u)^2$ in equation (2) can be neglected. Thus equation (1) can be simplified to

$$\frac{\dot{W}\sqrt{T}}{P_u} = (\alpha\gamma)_{id} A \sqrt{\left(\frac{2g}{nR}\right)\left(\frac{\Delta P}{P_u}\right)} \tag{3}$$

The flow coefficient α was shown in reference 1 to be a function of the radial clearance to knife-edge thickness ratio (δ/Δ) and the knife-edge thickness Δ, but to be independent of the Reynolds number above the critical value of 10^3. The lowest Reynolds number encountered in this investigation was above 10^3. Experimental curves of α as a function of (δ/Δ) as given by Egli are reproduced in figure 7. For an α value of 1.25 for the test seals (see fig. 1), a flow coefficient of 0.72 is obtained from figure 7.

The carryover factor γ was used by Egli to account for the increase in the leakage flow of a straight-through seal compared to the leakage flow of an ideal labyrinth seal. Egli defines an ideal labyrinth seal as one in which the kinetic energy of the leakage flow is completely destroyed following each throttling (radial gap between the knife-edge and the land). Thus by definition, the carryover factor γ is equal to 1.0 for an ideal seal. The performance of an ideal seal can be approached in an actual seal if the successive throttlings are well staggered in the radial direction with respect to each other. For a straight-through seal, where the successive throttlings are alined in the axial direction,

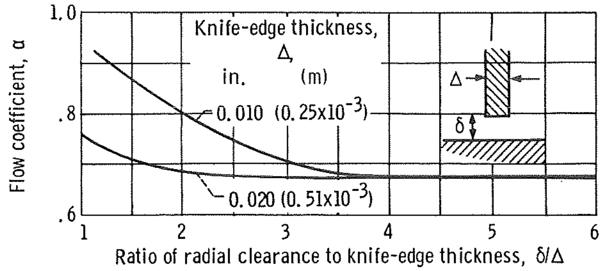


Figure 7. - Flow coefficient for labyrinth seals with sharp knife-edges (ref. 1).

and for a stepped seal with small or moderate step height, some of the kinetic energy will be carried over from one throttling to the next. This carryover of kinetic energy results in increased leakage flow through the seal. In this study, γ is assumed to apply to both test seals.

The leakage flow of an ideal seal was calculated from equation (3) using values of $\alpha = 0.72$, $\gamma = 1.0$, and flow area $A = 1.16$ square inches (7.5×10^{-4} m^2). The flow calculations were made both on the basis of the pressure difference across the center knife-edge ($n = 1$), and on the basis of pressure difference across the entire seal assembly ($n = 3$). For both of these calculations, γ was assumed to apply. (The average area of the three throttlings and the area of the center knife-edge throttling are equal for the straight-through seal; the difference in these two areas is negligible for the stepped seal. Therefore, the flow area of 1.16 in.^{2} (7.5×10^{-4} m^2) was used for all calculations.) For graphic presentation, the ideal seal results were correlated by comparing the weight flow parameter $\dot{W}\sqrt{T}/P_u$ (left side of eq. (3)), with the pressure parameter $\Delta P/P_u$ (right side of eq. (3)). The experimental results from the two test seals were correlated on the basis of these same two parameters.

The center knife-edge correlation is unconventional. But this correlation is of particular interest for the evaluation of the seal design application described in the appendix. The overall seal correlation is a more conventional approach, but would not be applicable for the evaluation of the design application under rotating conditions because of the unpredictable airflow and pressure influences previously mentioned. However, because of the conditions prevailing during the experimental phase of this study, these two correlations provided a dual evaluation of the leakage flow through the test seals. In these nonrotating tests, chambers A, B, C, and D (fig. 3) were affected only by the flow of air that leaked through the test seals; therefore both the center knife-edge pressure difference and the overall pressure difference provided meaningful monitors of seal leakage.

There were two advantages in measuring the pressure difference across the overall seal assembly as compared to measuring the pressure difference across the center knife-edge. For the same leakage flow, the overall pressure difference was about three times as large as the pressure difference measurements obtained across the center knife-edge. A fixed systematic error in the measurement would result in a percentage error of about one-third for the overall pressure difference as compared to the center knife edge pressure difference. For a stationary test, where the rotational and other disrupting effects are absent, the larger size of chambers A and B relative to chambers C and D (fig. 3) are also conducive to more accurate measurements because the larger chambers are less likely to be affected by slight velocity and pressure fluctuations.

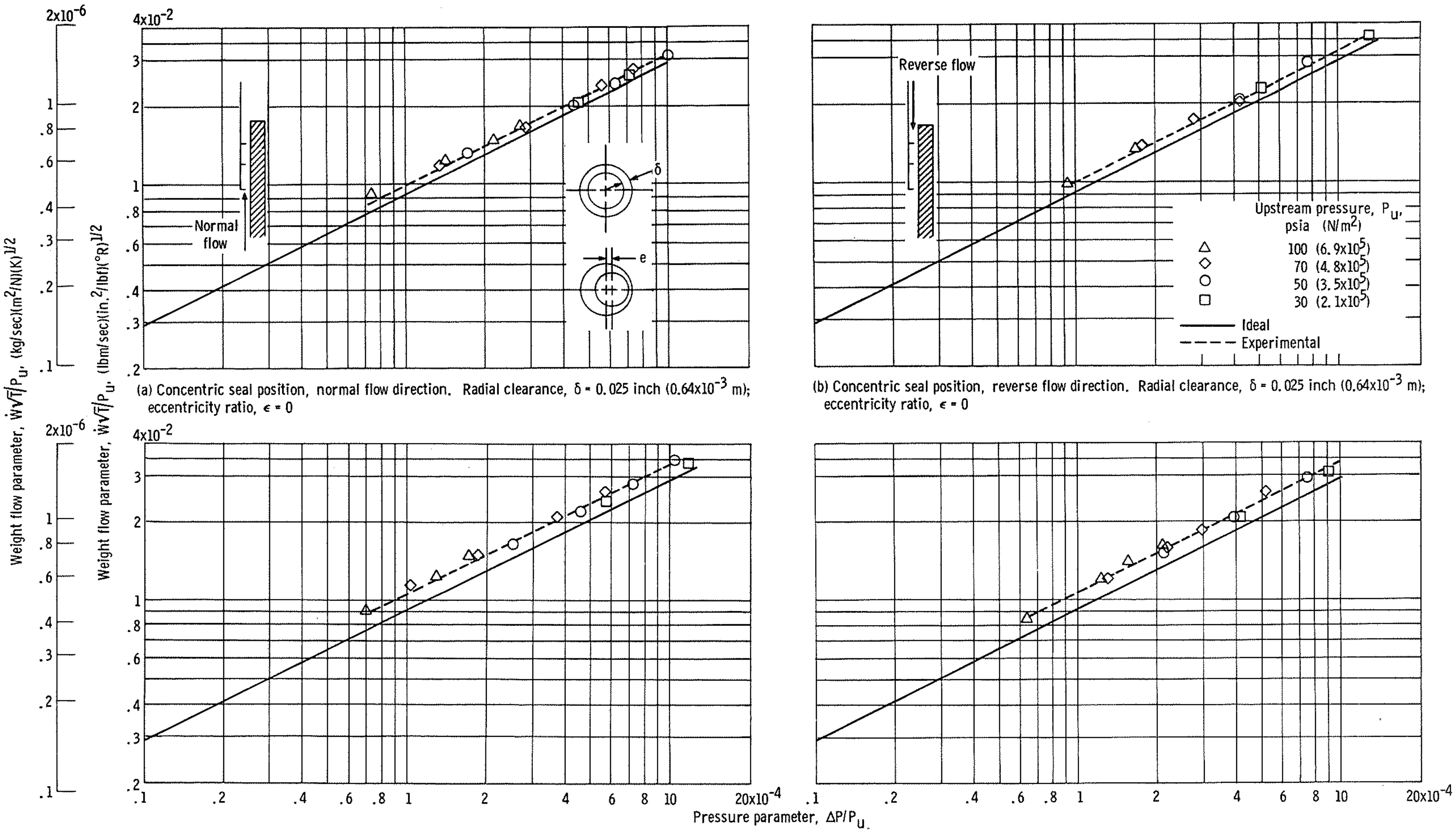
RESULTS AND DISCUSSION

The experimental data and the calculated ideal seal results of this investigation are shown in figures 8 to 11. Results of the straight-through test seal are presented in figures 8 and 9; results for the stepped test seal are presented in figures 10 and 11. In each figure, the weight flow parameter $\dot{W}\sqrt{T}/P_u$ is shown as a function of the pressure parameter $\Delta P/P_u$ for various geometric combinations of seal configuration, seal position, and leakage flow direction. Experimental data are indicated by individual data points, with a dashed line to indicate the average results. The ideal seal results are indicated by solid lines. For each combination of seal geometry, the data are correlated for both center knife-edge pressure difference and the overall seal pressure difference.

Examination of figures 8 to 11 showed that on each data plot the slopes of the lines through the experimental and the ideal seal leakage results are equal to each other. These agreements in the slopes of the lines indicated that, for each combination of seal geometry, the experimental and the ideal seal leakage flows vary only by a constant factor. The significance of this constant factor can be seen by referring to equation (3) which was used to calculate the ideal seal leakage flows.

The only quantities appearing in equation (3) that cannot be measured directly and assigned numerical values from the experimental data are α and γ. Therefore, the constant difference between the experimental and ideal leakage values can be directly attributed to the effect of α and γ, and is equal to the ratio of $(\alpha\gamma)_{exp}/(\alpha\gamma)_{id}$. This ratio will be referred to hereinafter as the $\alpha\gamma$ ratio.

A summary of the $\alpha\gamma$ ratios determined from comparisons of the ideal and experimental lines on figures 8 to 11 are presented in table II. From the definition of the $\alpha\gamma$ ratio, the value is 1.0 for an ideal seal. However, values less than 1.0 can be obtained for this ratio, as shown in table II. This condition can exist because of one or both of the following: (1) the assumed value of α_{id} is not correct for the particular seal; (2) some systematic error is present in the experimental data. As shown from the data plots, the leakage flow for a labyrinth seal with geometries similar to the test



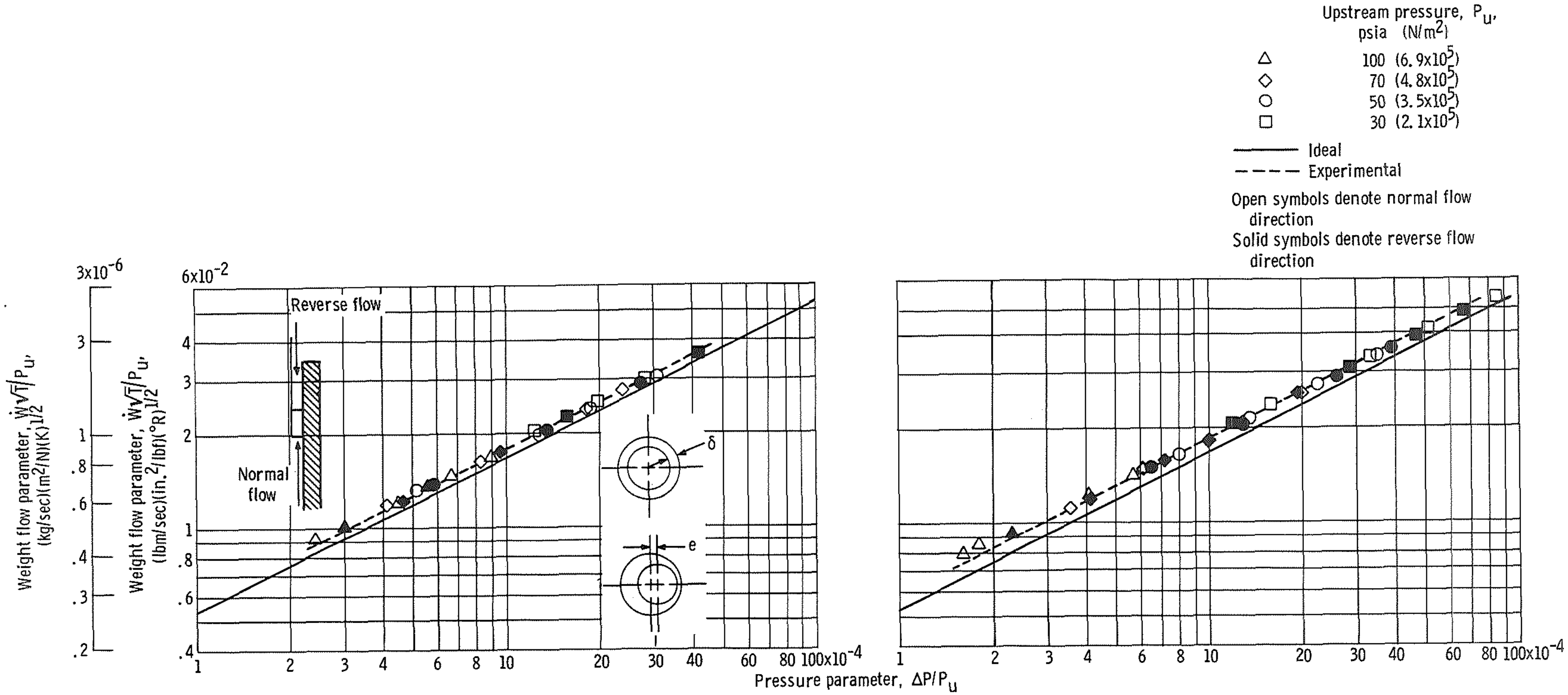
(a) Concentric seal position, normal flow direction. Radial clearance, δ = 0.025 inch (0.64x10^{-3} m); eccentricity ratio, ε = 0

(b) Concentric seal position, reverse flow direction. Radial clearance, δ = 0.025 inch (0.64x10^{-3} m); eccentricity ratio, ε = 0

(c) Eccentric seal position, normal flow direction. Eccentricity ratio, ε = 0.72.

(d) Eccentric seal position, reverse flow direction. Eccentricity ratio, ε = 0.72.

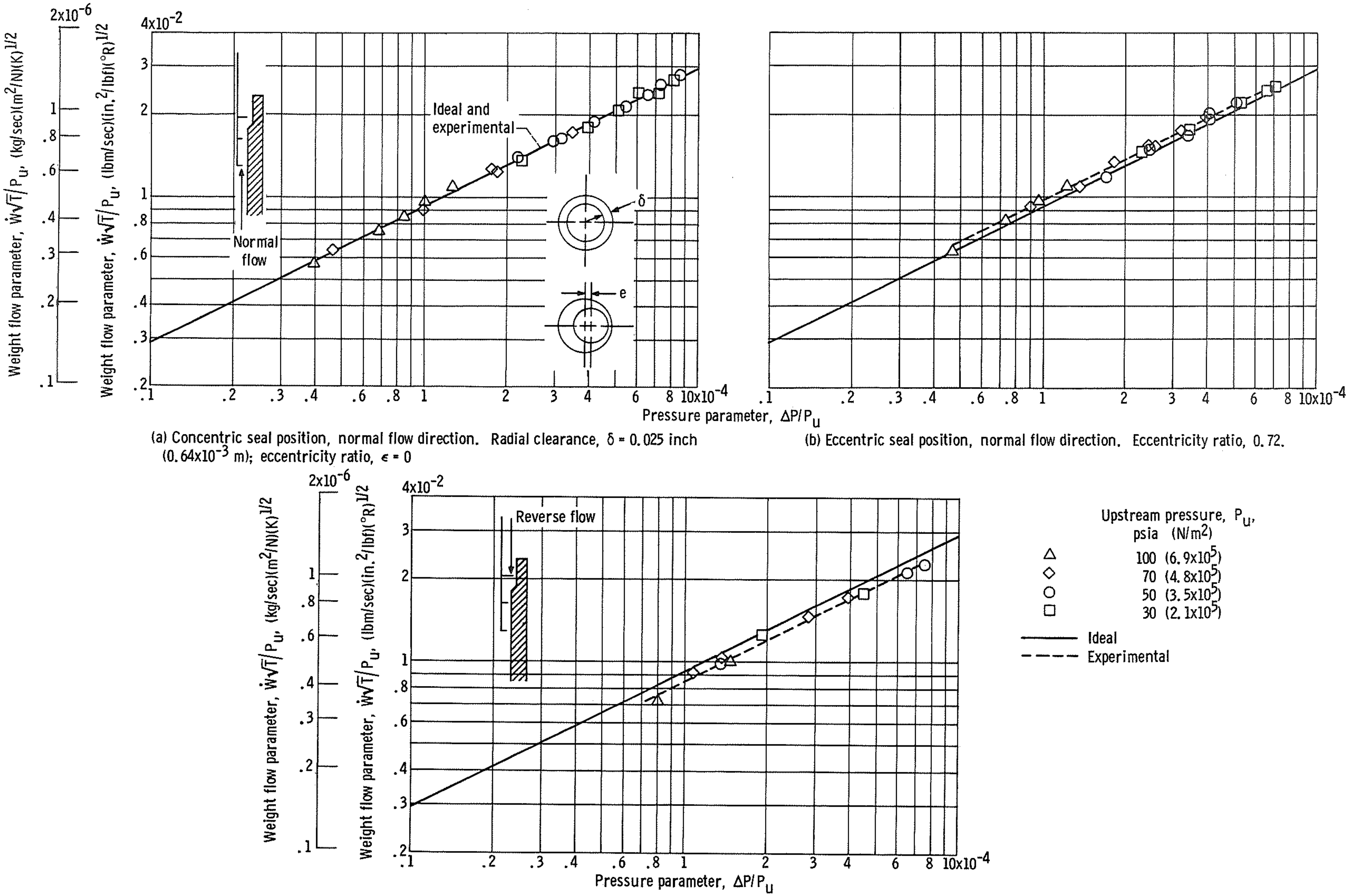
Figure 8. - Center knife-edge pressure difference correlation of weight flow through three-stage straight-through labyrinth seal.



(a) Concentric seal position, normal and reverse flow directions. Radial clearance, δ = 0.025 inch (0.64x10^{-3} m); eccentricity ratio, ϵ = 0

(b) Eccentric seal position, normal and reverse flow directions. Eccentricity ratio, ϵ = 0.72.

Figure 9. - Overall pressure difference correlation of weight flow through three-stage straight-through labyrinth seal.

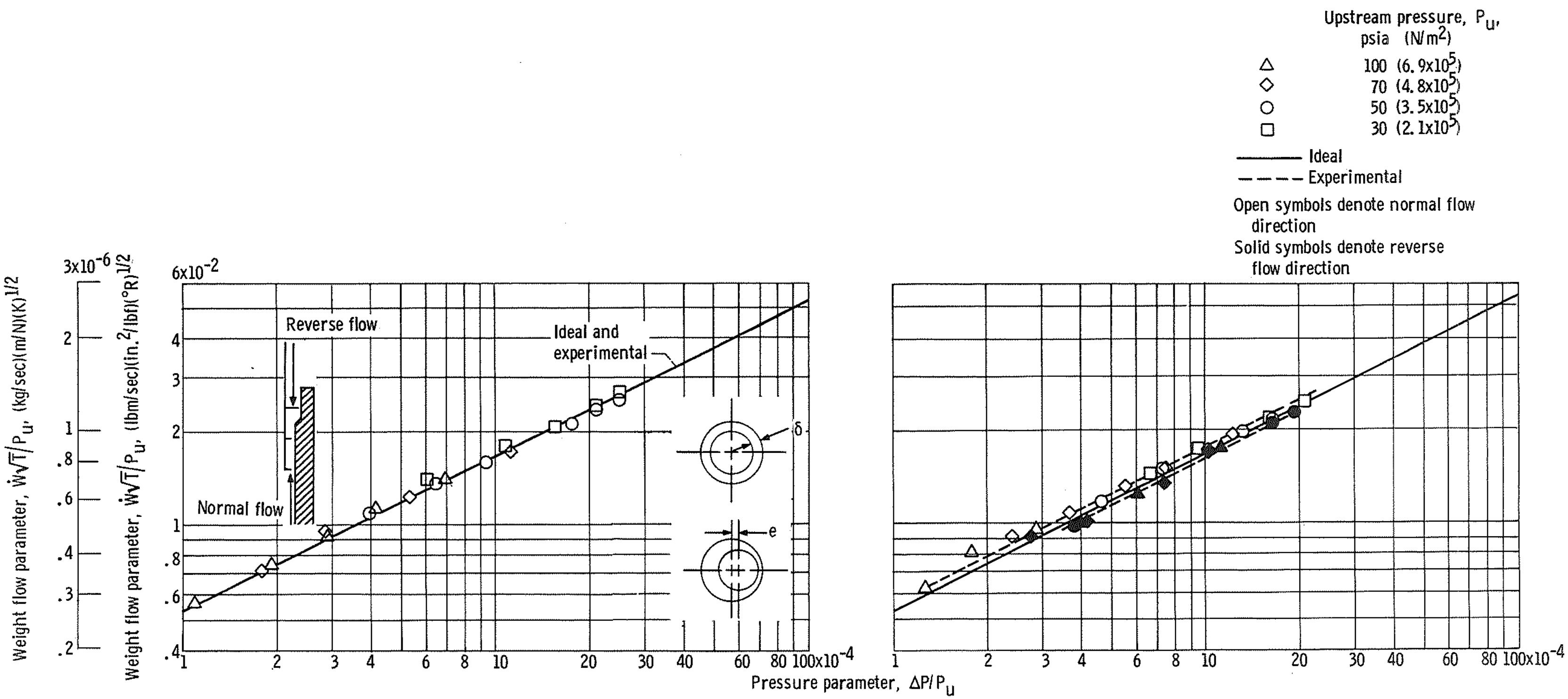


(a) Concentric seal position, normal flow direction. Radial clearance, δ = 0.025 inch (0.64x10^{-3} m); eccentricity ratio, ϵ = 0

(b) Eccentric seal position, normal flow direction. Eccentricity ratio, 0.72.

(c) Eccentric seal position, reverse flow direction. Eccentricity ratio, 0.72.

Figure 10. - Center knife-edge pressure difference correlation of weight flow through three-stage stepped labyrinth seal.



(a) Concentric seal position, normal flow direction. Radial clearance, δ = 0.025 inch (0.64×10^{-3}); eccentricity ratio, ε = 0

(b) Eccentric seal position, normal and reverse flow directions. Eccentricity ratio, ε = 0.72.

Figure 11. - Overall pressure difference correlation of weight flow through three-stage stepped labyrinth seal.

TABLE II. - SUMMARY OF EXPERIMENTALLY DETERMINED $\alpha\gamma$ RATIO

Item	Type of test seal	Type of ΔP measurement	Seal position	Eccentricity ratio, ϵ	Flow direction	$\alpha\gamma$ ratio, $\frac{(\alpha\gamma)_{exp}}{(\alpha\gamma)_{id}}$	Figure
1	Straight-through	Across center knife-edge	Concentric	0	Normal	1.08	8(a)
2			Concentric	0	Reverse	1.10	8(b)
3			Eccentric	.72	Normal	1.15	8(c)
4			Eccentric	.72	Reverse	1.16	8(d)
5		Across entire seal assembly	Concentric	0	Normal	1.07	9(a)
6			Concentric	0	Reverse	1.07	9(a)
7			Eccentric	.72	Normal	1.11	9(b)
8			Eccentric	.72	Reverse	1.11	9(b)
9	Stepped	Across center knife-edge	Concentric	0	Normal	1.01	10(a)
10			Eccentric	.72	Normal	1.05	10(b)
11			Eccentric	.72	Reverse	.93	10(c)
12		Across entire seal assembly	Concentric	0	Normal	1.01	11(a)
13			Eccentric	.72	Normal	1.06	11(b)
14			Eccentric	.72	Reverse	.99	11(b)

seals can be predicted by multiplying the ideal leakage flows (determined by eq. (3)) by the experimentally determined $\alpha\gamma$ ratio listed in table II.

The relative comparisons of the values of the $\alpha\gamma$ ratio for the various combinations of seal geometry afford a ready means of judging relative seal performance for these combinations. These comparisons are made on the basis of tabulated values of $\alpha\gamma$ ratio in table II, where a change of 0.01 in this ratio is defined as a change of 1 percent. Such comparisons are made below to illustrate the effects of flow direction and eccentricity for the straight-through and the stepped seals. In the comparisons for the test seal data, the overall seal pressure difference correlations were considered to be more reliable than the center knife-edge correlations under static (nonrotating) conditions because of the factors discussed previously.

Straight-Through Seal

Effect of flow direction. - Because of the geometric symmetry of the straight-through seal (fig. 1(a)), no major effect due to change of flow direction would be anticipated. An experimental evaluation of the effect of flow direction can be made from comparisons of the tabulated values of the $\alpha\gamma$ ratios in table II. For the center knife-edge correlation of the concentric seal position, the $\alpha\gamma$ ratio (and therefore the leakage flow at a given value of $\Delta P/P_u$) is 2 percent larger for the reverse flow direction than for the normal flow direction (items 1 and 2 in table II). A like comparison for the eccentric seal position showed a 1 percent larger value of this ratio for the reverse flow direction than for the normal flow direction. When the experimental results were correlated on the pressure difference across the entire seal assembly, no change in leakage flow with flow direction was noted either for the concentric seal position (items 5 and 6, table II) or for the eccentric seal position (items 7 and 8, table II).

Thus, the experimental results of this study confirm that the effect of flow direction on the leakage flow rate of the straight-through test seal was indeed very small.

Effect of eccentricity. - The effect of eccentricity on the leakage flow of the straight-through test seal also can be determined from comparisons of the $\alpha\gamma$ ratios tabulated in table II. For the center knife-edge correlation, the value of this ratio for the concentric seal position with flow in the normal direction is 1.08 (item 1). For the eccentric seal position, an increase in this ratio of 7 percent to a value of 1.15 is noted (item 3). This increase in $\alpha\gamma$ ratio indicated a corresponding increase in leakage flow. For the reverse flow direction, the $\alpha\gamma$ ratios were 1.10 for the concentric position (item 2) and 1.16 for the eccentric position (item 4). This is an increase of 6 percent. The trend of increased leakage flow with eccentricity is also confirmed by the results as correlated for the overall seal pressure difference.

The $\alpha\gamma$ ratios for both normal flow direction (items 5 and 7) and reverse flow direction (items 6 and 8) increased 4 percent when the seal position was changed from concentric to eccentric.

It can be concluded from these comparisons that a significant increase in leakage flow occurred for the straight-through seal when the seal position was changed from concentric to eccentric. An increase in leakage flow due to eccentricity was also reported in reference 5 for flow through fine annular clearance and with smooth walls, such as those in journal bearings.

Stepped Seal

Effect of flow direction. - The effect of flow direction on the leakage flow through the stepped test seal can be illustrated by comparison of the $\alpha\gamma$ ratio value in table II. For the eccentric seal position, a decrease of 12 percent in the $\alpha\gamma$ ratio (and therefore the leakage flow) was indicated for the center knife-edge correlations when the flow direction was changed from normal (item 10) to reverse (item 11). For the overall seal correlations, this change in flow direction resulted in an indicated decrease of 7 percent in the $\alpha\gamma$ ratio (items 13 and 14).

A decrease in leakage flow with change in flow direction, as indicated by both correlations, can be attributed to the increased friction loss for airflow against the step (reverse flow) compared to flow over the step (normal flow). The apparent discrepancy in the magnitude of the flow decrease with change in flow direction as indicated by the two correlations is probably due to the extreme sensitivity of the sensed pressure in chamber C (fig. 4(a)) to the nature of the friction loss caused by the step. Because of this uncertainty and the other effects on the accuracy of measurement mentioned in the section PROCEDURES, the overall correlation is considered to be more accurate than the center knife-edge correlation.

Effect of eccentricity. - The effects of eccentricity on leakage flow through the stepped test seal can be determined by comparison of the $\alpha\gamma$ ratios for items 9 and 10 in table II (center knife-edge correlations) and for items 12 and 13 (overall seal correlations). The first of these comparisons showed a 4 percent increase in the $\alpha\gamma$ ratio (and in leakage flow) when the seal position was changed from an eccentricity ratio of 0 to 0.72. The second comparison (overall correlations) showed a 5 percent increase for the same change in seal position. The magnitude of this increase is about the same as the increase in leakage flow caused in the straight-through test seal with similar change in seal position.

SUMMARY OF RESULTS

The following results were obtained from a study of straight-through and stepped labyrinth seals, under nonrotating conditions at very small pressure differences:

1. The analytical equation developed by Egli can be used for predicting leakage flows through labyrinth seals for the very small pressure differences covered in this investigation, if the product of the flow coefficient and the carryover factor are known.

2. For the straight-through seal, the leakage flow was insensitive to the direction of airflow. For the stepped seal, the leakage flow based on the overall seal pressure difference was 7 percent greater when the air was flowing over the step than when the air was flowing against the step.

3. The effect of eccentricity was to increase the leakage flow through the labyrinth seals. When the eccentricity ratio of the seal was changed from 0 to 0.72, the leakage flow increased about 5 percent for both the straight-through and the stepped seal, based on the overall seal pressure difference.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, October 30, 1969,
720-03.

APPENDIX - ENGINE SEAL SYSTEM

The cooling air seal system for a turbine cooling research engine generated the necessity for the seal leakage study reported herein. This seal system is shown schematically in figure 2. The seal system is used for the transfer of cooling air from stationary to rotating parts of the engine. Cooling air is supplied to a five-blade rotating cascade that is part of the turbine blading. This rotating cascade is cooled independently of the remaining blades on the turbine disk. Heat-transfer evaluations for the experimental air-cooled blades that will be mounted in this rotating cascade required that the quantity of blade cooling air be known accurately. Precise metering of this airflow could be done on the stationary part of the cooling air supply system. However, the precision of this measurement could be negated in transferring this metered air to the rotating part of the system if a good seal were not used at the transfer point. Balanced-pressure labyrinth seals were chosen for this engine seal application for the following reasons:

(1) The high-temperature environment (1200° F (650° C) cooling air temperature) precluded the use of carbon-type seals

(2) The large relative motions that occur between seal parts due to differential thermal growth would make the maintenance of proper alinement of any face-type seals difficult

(3) The state-of-the-art for labyrinth seal design was further advanced than for other types of seal designs

The use of concentric seals to form an annular air transfer chamber (chamber 4, fig. 2) was dictated by the design features of the research engine. These design features of the research engine. These design features fixed the minimum diameter of the seal system at 3.6 inches (0.09 m). The outer diameter of the seal system was held to the smallest value practical to minimize seal leakage. The net result of this limited space envelope for the transfer of cooling air was a design with extremely large peripheral seal lengths compared to the flow area of the air chamber which was enclosed by the seals.

The flow path of the test blade cooling air for the research engine is as follows. Cooling air for the turbine blades is ducted into annular chamber 4 (fig. 2) on the stationary (nonrotating) part of the seal system. From this chamber, the cooling air flows through several radial holes in the rotating part of the seal and then to the blades. Leakage of cooling air from chamber 4 is minimized by establishing pressure balances between chamber 4 and the annular pressure-balance chambers 1 and 7 in figure 2. Separately controlled air sources are supplied to each of these pressure-balance chambers. Chambers 1 and 7 are separated from the blade cooling air chamber (4) by three-stage straight-through labyrinth seals. The three knife-edges and the smooth land of each seal form two identical chambers (2 and 3 on the outer radius and 5 and 6 on the inner radius). In each seal, adjacent identical chambers are separated from each other by the center knife-edge of the seal. The existence of a balance pressure condition (and, therefore, a condition of no leakage flow) is detected by sensing a zero pressure difference across the center knife-edge by pressure taps located as indicated by the solid dot symbols in figure 2.

The dimensions of the test seals were chosen originally to be representative of the anticipated dimensions of the engine seal system. However, because of space limitations in the engine, the dimensions for land diameter and knife-edge spacing in the final design of the engine seal system were reduced from the corresponding dimensions used for the test seals. The final engine seal design used only straight-through seals in the positions where pressure difference monitoring was to be done. This choice was made before the conclusion of the present study. The significant dimensions of the test seals (fig. 1) and the engine seals for which the test seals served as models (inner and outer seals on fig. 2) are given in table III.

Two-stage stepped labyrinth seals were used at the other boundaries of the annular pressure-balance chambers to limit the quantity of air required to establish pressure balances between the balance chambers 1 and 7 and the cooling air chamber 4.

TABLE III. - SEAL DIMENSIONS

Measurement	Test seal dimension		Engine seal dimension	
	in.	m	in.	m
Nominal land diameter, D	14.8	376×10^{-3}	[a]4.4 [b]6.0	[a]110×10^{-3} [b]150×10^{-3}
Knife-edge spacing, s	.75	19×10^{-3}	.375	9.5×10^{-3}
Knife-edge thickness, Δ	.02	$.51 \times 10^{-3}$	.02	$.51 \times 10^{-3}$
Radial clearance (between knife-edge and land), δ	.025	$.64 \times 10^{-3}$	.025	$.64 \times 10^{-3}$
Step height (stepped seal only), h	.05	1.3×10^{-3}	-------	------------

[a]Inner seal.
[b]Outer seal.

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